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IMPROVEMENTS IN DRY PUMPS

This invention relates to dry pumps, more specifically to Roots, Northey (or "claw") and screw pumps which are typically used in vacuum applications. The invention is directed to improvements in the operation of the aforementioned pumps.

Dry pumps are widely used in industrial processes to provide a clean and/or low pressure environment for the manufacture of products. Applications include the pharmaceutical and semiconductor manufacturing industries. Such pumps include an essentially dry (or oil free) pumping mechanism, but generally also include some components, such as bearings and transmission gears, for driving the pumping mechanism and which require lubrication in order to be effective.

Dry pumps incorporating Roots and/or Northey mechanisms are commonly multi-stage positive displacement pumps employing intermeshing rotors in each vacuum chamber. The rotors may have the same type of profile in each chamber or the profile may change from chamber to chamber.

A typical screw pump mechanism comprises two parallel spaced shafts each carrying externally threaded rotors, the shafts being mounted in a pump body such that the threads of the rotors intermesh. Close tolerances between the rotor threads at the points of intermeshing and with the internal surface of the pump body (which acts as a stator), causes volumes of gas entering at an inlet to be trapped between the threads of the rotors and the internal surface and thereby urged towards an outlet of the pump as the rotors rotate. Various adaptations of the basic screw pump mechanism are known, for example, there exist screw pumps with variable pitch screw threads and/or mechanisms wherein the height (or outside diameter) of the screw thread tapers decreasingly in a direction from the pump inlet to the exhaust of the pump. In the latter case, the rotors are mounted in a tapering bore of the stator.

It is desirable when operating a dry pump to achieve a desired pressure ("ultimate pressure"), which is typically significantly below atmospheric pressure, that the input power needed to operate the pump is minimised. The size of the pump exhaust volume has a considerable effect on the input power needed to operate a pump at ultimate pressure. The input power can be maintained low at ultimate pressure by inbuilding a high volume ratio between the inlet volume and the exhaust volume of the pump. A disadvantage of this arrangement is that as the inlet pressure of the pump increases towards atmospheric pressure, there is a significant increase in the input power requirements of the pump.

In the prior art, high pump internal pressures have been avoided by inclusion of a blow-off valve within the pump, which can be activated to release pressure and prevent build up of excessive pressure in the pump. In some situations, the performance of these valves can be adversely affected by the build-up of process media on or near sealing surfaces, reducing the efficiency with which pressure build up can be relieved.

In accordance with one aspect of the present invention, there is provided a dry pump comprising a stator housing first and second intermeshing rotors adapted for counter-rotation within the stator, and means for effecting axial movement of the rotors within the stator to vary at least one clearance between the rotors and the stator during use of the pump.

Actively controlling the axial position of the rotors within the stator in real time in response, for example, to operational conditions within the pump allows a clearance between the rotors and the stator to be increased, decreased or maintained at a constant level as necessary during use of the pump. For example, the rotor to stator clearance can be increased or even maximised when the pump is switched off following pumping of sticky or dusty atmospheres, so as to prevent problems occurring upon restart. The axial clearance between rotor and stator can fill with process deposits during operation, and when the pump is stopped the rotors will cool and shrink on to the process deposits, potentially

locking the pump. Moving the rotors relative to the stator can allow the axial clearance to increase when the pump is stopped, thereby increasing the likelihood of the pump restarting. Once the pump has started, the axial clearance can then be reduced back to a normal running clearance during operation of the pump.

Alternatively, or additionally, the rotors can be actively moved closer to the stator surface during use of the pump to scrape off process media built up on the stator surface.

The rotor to stator clearance may also be controlled to optimise pump performance for different pumped gas species. For example, the clearance can be decreased or increased when pumping hydrogen or an inert gas such as argon so as to achieve optimum performance without pump seizure.

This movement of the rotors within the stator can significantly reduce the effect of operational variables, such as backpressure, running temperatures and gas type, on pump performance. Furthermore, the ability to actively control rotor position can relax the manufacturing precision of components. This can bring a significant reduction in cost due to the potential removal of grinding operations and the reduction in scrap levels.

For example, in one embodiment the pump comprises means for effecting axial movement of the rotors in response to an axial load generated in the rotors during operation of the pump. When in operation, internal pressure within a pump produces an axial thrust load in the rotor. This thrust load is proportional to the amount of gas compression work being performed by the pump and hence the input power requirements of the pump. For example, the efficiency of gas compression of a screw pump is, to a large extent, dictated by the clearance between the internal surface of the stator which carries the screw threaded rotors and the rotors themselves. Where the rotors are tapered, they may be moved both simultaneously and synchronously away from the stator face effectively increasing the radial clearance, reducing the compression and hence the power

input requirements. Similarly, tapered Roots rotors may be moved both simultaneously and synchronously away from the stator face effectively increasing the radial clearance, reducing the compression and hence the power input requirements.

Each rotor is typically mounted on, or integral with, a respective shaft rotatably mounted within the pump, the pump comprising a bearing assembly for rotatably supporting the shafts relative to the stator. In preferred embodiments, the means for effecting axial movement of the rotors comprises means for moving the bearing assembly relative to the stator. For example, in one embodiment, the bearing assembly is free to move in an axial direction within a housing, the means for effecting axial movement of the rotors comprising a spring mechanism arranged with respect to a rotor such that when the rotor is subjected to an axial load, the spring mechanism compresses or extends causing an axial reactive load.

When an axial load generated in a rotor tends to cause axial displacement of the rotor and bearing assembly, the spring may be compressed or extended (depending on its position). Assuming the load does not exceed the elastic limit of the spring, the spring will react to vary the axial position of the rotor. By selecting a spring with a suitable spring constant, the arrangement can be used to vary the rotor to stator clearance giving a relatively constant level of gas compression work over a wide range of inlet pressures, thereby moderating the power input requirements of the pump.

In other embodiments, the pump comprises control means for actively controlling operation of the means for effecting axial movement of the rotors. For example, a piston or other actuator may be provided for moving the rotors, the control means controlling movement of the actuator to control the axial position of the rotors. In preferred embodiments, the control means controls a motor adapted to rotate a drive shaft that engages the actuator so as to axially move the actuator relative to the stator with rotation of the drive shaft. The drive shaft may, for example, comprise a lead screw passing through a conformingly-threaded aperture in the

actuator. Alternatively, the control means may comprise one or more electromagnets for moving the actuator.

Any other convenient mechanism for accurately moving the rotors relative to the stator may be provided. For example, a piezoelectric actuator may be provided, which deforms in response to a voltage supplied thereto by the control means to move the rotors within the stator. Alternatively, a metallic ring, tube or other element can be provided, which is selectively heated by the control means so that the resulting thermal expansion of the element causes the rotors to move within the stator. The most appropriate mechanism can be chosen for the extent of the required movement of the rotors relative to the stator. For example, for Northey rotors, the maximum required movement may be less than 100 microns, whereas for tapered screw rotors the required movement may be around 1 mm.

Where movement of the rotors is effected by movement of the bearing assembly relative to the stator, the actuator may conveniently comprise part of, or is carried by, a housing for the bearing assembly. The housing for the bearing assembly preferably carries an internal sealing mechanism for the pump. This bearing assembly preferably supports one end of each shaft, with a second bearing assembly being provided for supporting the other end of each shaft. This bearing assembly may be fixed relative to the stator or may be arranged to move with the shafts. A housing for this latter second bearing assembly may also carry an internal sealing mechanism for the pump. This can ensure that movement of the shafts relative to the stator does not compromise the internal pump sealing.

One or both of the housings for the bearing assemblies may define an end surface of the stator so that the end surface moves with the rotors as the axial position of the rotors is adjusted, thereby avoiding collision between the ends of the rotors and the end surfaces of the stator by maintaining a constant clearance between the ends of the rotors and the end surfaces of the stator.

The control device may be configured to receive a signal indicative of an operational parameter of the pump, and to control the axial position of the rotors in dependence of this signal. The operational parameter may include one of the temperature of and/or within the stator, ultimate vacuum calibration at start up, backpressure, exhaust temperature, power consumption and inlet pressure. In relation to temperature, cooling water upsets could create a problem in that thermal shocking can distort the stator to such an extent that the rotor makes contact with the stator. By measuring the stator temperature and incoming water temperature, it is possible to detect the onset of a seizure condition and protect the pump by increasing the rotor to stator clearance as thermal shock occurs. High backpressure results in an increase in internal gas temperature and rotor to stator differential, which could lead to pump seizure. By measuring the internal gas temperature it is possible to modify the rotor to stator clearance to accommodate the increase in backpressure. In relation to power consumption, the control device may be configured to receive a signal indicative of the power consumption of a motor for rotating the rotors, and to control actuation of the actuator in response thereto.

Alternatively, or in addition, the control means may comprise a sensor for detecting the size, or the rate of change, of the clearance between the rotors and the stator, and is configured to control the means for effecting axial movement of the rotors in response to an output from the sensor. The sensor may be conveniently provided by a Hall effect sensor. The sensor can be calibrated by determining the position of the rotors within the pump when there is zero axial clearance between the rotors and the stator. This could be achieved by moving the rotors axially until contact occurs (with the rotors non-rotational) either before use or once the pump has warmed up in order to account for thermal effects. Alternatively, these thermal effects could be built into the control means, so that they are taken into account when determining the size of the axial clearance from a signal output from the sensor.

The means for effecting axial movement of the rotors is preferably arranged so as to ensure both rotors are maintained in the same axial position, but may also be configured so as to permit relative axial movement between the rotors. Typically, such relative movement will be within the limits of rotor contact and might be used with the rotors in operation to scrape off process media build up on the flanks of the rotors or to allow fine tuning of the timing. The relative movement between the rotors can be achieved using independent means for effecting axial movement of each rotor, for example respective actuator arrangements as previously mentioned. An associated control device may be configured to actuate movement of the rotors independently of one another. This could be done while the rotors are stationary by monitoring the achievable travel of a rotor, or while operating at full speed by monitoring shaft torque or motor current.

Where the rotors have intermeshing screw threads, at least part of the screw threads preferably has an outside diameter that tapers decreasingly in a direction from the pump inlet to the exhaust of the pump. In one embodiment, each screw thread has a diameter that gradually decreases from the pump inlet to the exhaust. In another embodiment, only part of the screw thread of each rotor has an outside diameter that tapers towards the exhaust of the pump, the remainder of the screw thread having a substantially constant diameter. There are a number of advantages particularly associated with this latter embodiment. Firstly, vacuum pump exhaust gas temperatures vary with running conditions, and have an effect on the rotor to stator clearance at the exhaust (low vacuum) end of the pump. Control of the rotor to stator clearance in the exhaust stages allows the optimisation of performance and power consumption. The inlet (high vacuum) temperature does not vary as considerably as the exhaust and hence rotor to stator control in the inlet stages is of lesser importance. Secondly, during roughing (pumping large volumes of gas at or near atmospheric pressure) performance can be optimised by bypassing the low vacuum stages of the pump. The rotor to stator clearance in the exhaust stages can be increased to act as a pressure relief valve, with the rotor to stator clearance at the inlet stages remaining constant so as to maximise pumping efficiency. Where the rotors are, at least in

part, tapered, the size of the radial clearance between the rotors and the stator can be determined from the size of the axial clearance between the rotors and the stator. The relationship between the axial and the radial clearances can be established during testing.

The rotors may be pulsed between two axial positions to remove process deposits in the axial clearances between the rotors and the stator. For example, the control device may be configured to move the rotors at a first speed in one axial direction to increase the axial clearance between the rotors and the stator, and to move the rotors at a second speed different from the first speed to decrease the axial clearance between the rotors and the stator. In order to prevent tripping of the pump motor, the rate of decrease of the axial clearance is preferably greater than the rate of increase of the axial clearance. A linear encoder may be provided to prevent seizure at the extremities of the axial positions of the rotors.

In a second aspect, the present invention provides a method of controlling operation of a pump comprising a stator housing first and second intermeshing rotors adapted for counter-rotation within the stator, the method comprising the steps of axially moving the rotors relative to the stator to increase an axial clearance between the rotors and the stator when rotors are stationary, subsequently starting rotor rotation, and, during rotor rotation, axially moving the rotors relative to the stator to decrease the axial clearance between the rotors and the stator.

The method may further comprise subsequently, and preferably repeatedly, increasing and decreasing the axial clearance during use of the pump to remove deposits from the axial clearance. Thus, in a third aspect the present invention provides a method of controlling operation of a pump comprising a stator housing first and second intermeshing rotors adapted for counter-rotation within the stator, the method comprising the steps of sequentially axially moving the rotors in opposite directions relative to the stator to periodically vary an axial clearance between the rotors and the stator to remove deposits from the axial clearance.

Features described above in relation to the first aspect of the invention are equally applicable to the method aspects of the invention, and vice versa.

By way of example, some embodiments of the invention will now be further described with reference to the following Figures in which:

Figure 1 is a side view of a first embodiment of a screw pump;

Figure 2 is a plan view of a pump of Figure 1;

Figure 3 shows a section through the plane B-B marked in Figure 1;

Figure 4 shows a section through the plane A-A marked in Figure 2;

Figure 5 is a plan view of means for effecting axial movement of the rotors in the pump of Figure 1;

Figure 6 shows a section through the plane A-A marked in Figure 5;

Figure 7 is a perspective view of the means for effecting axial movement of the rotors in the pump of Figure 1;

Figure 8 shows a section through a second embodiment of a screw pump;

Figure 9 shows a section through a third embodiment of a screw pump;

Figure 10 shows a section through an embodiment of a Northey pump; and

Figure 11 shows a section through an embodiment of a Roots pump.

With reference first to Figures 1 to 4, the pump 10 includes a pump body 12 having a first part 14 and a second part 16 defining a pumping chamber 18. A

fluid inlet 20 to the chamber 18 is formed in the first part 14 of the pump body 12, and a fluid outlet 22 from the chamber 18 is formed in the second part 16 of the pump body 12.

The pump 10 further includes a first shaft 24 and, spaced therefrom and parallel thereto, a second shaft 26. Bearings 28 are provided for supporting the shafts 24, 26. The shafts 24, 26 are adapted for rotation about the longitudinal axes thereof in a contra-rotational direction. One of the shafts 24 is connected to a drive motor 30 via a drive mechanism 32, the shafts being coupled together by means of timing gears so that in use the shafts 24, 26 rotate at the same speed but in opposite directions.

A first rotor 34 is mounted on the first shaft 24 for rotary movement within the chamber 18, and a second rotor 36 is similarly mounted on the second shaft 26. Each of the two rotors 34, 36 has a tapered shape and has a helical vane or thread 38, 40 respectively formed on the outer surface thereof, the threads intermeshing as illustrated.

In this embodiment, the screw threads 38, 40 of the rotors 34, 36 have an outside diameter which tapers decreasingly in a direction from the inlet 20 to the outlet 22 of the pump 10, and the inner surface of the pumping chamber 18, which acts as a stator during use of the pump, conformingly tapers towards the pump outlet 22. The shape of the rotors 34, 36 and in particular the shapes of the threads 38, 40 relative to each other and to the inner surface of the pumping chamber 18, are calculated to ensure close tolerances with the inner surface of the pumping chamber 18.

In order to control the axial position of the rotors 34, 36 within the chamber 18, the pump 10 includes bearing assemblies 42 each slidably mounted within a respective cylindrical housing 44, as shown in more detail in Figures 5 to 7, located at the exhaust end of the pump 10. Each cylindrical housing 44 is

fastened to a respective shaft 24, 26, the cylindrical housings 44 being connected to each other by means of a connecting arm 46.

Each bearing assembly 42 comprises a pair of angular contact bearings 48 arranged in a back to back configuration to maintain the lateral position of the shaft 24 passing through the bearing assembly 42 with respect to the pump body 12 whilst allowing axial movement of the shaft and rotation of the shaft about its longitudinal axis. A small clearance is provided between the outer surface of the bearings 48 and the inner surface of the cylindrical housing 44. The cylindrical housing 44 also has a small axial clearance with the pump body 12 which allows the initial axial clearance to be fixed during pump assembly by placing shim material between the pump body and the clamping flange 50 of each cylindrical housing 44.

Located between the bearings 48 and the end wall 52 of the cylindrical housing 44 are a spacer ring 54 and a spring 56. The bearings 48 are retained in the housing 44 by means of a clamping ring 58 such that a preload on the spring 56 is set for the running load condition (and input power of the pump). The end wall 52 extends radially inwardly towards a collar 60 which forms part of an assembly for fastening the cylindrical housing 44 to the shaft.

In use, compression work done by the pump 10 results in an axial load tending to move the rotors 34, 36 in a direction from the outlet 22 towards the inlet 20. The axial load acts against the springs 56 to cause the rotors 34, 36 to move in an axial direction, thereby changing the radial clearance between the threads 38, 40 of the rotors 34, 36 and the stator in proportion to the axial load. By changing the characteristic spring rate, the input power of the pump can be tailored to a specific application over the speed range of the pump. Where there is any difference in axial load on the rotors 34, 36, the connector 46 rigidly connecting the two cylindrical housings 44 together ensures that both rotors are repositioned simultaneously, avoiding any interference which may occur between the threads

38, 40 of the rotors 34, 36 should they become misaligned with respect to each other.

In a variation of the embodiment of Figures 1 to 7, the axial positions of the rotors are controlled by pneumatically controlled pistons. Movement of the pistons may be controlled by a control mechanism which may include a force sensor which detects the axial load on a given rotor axis and causes a reactive force to be applied by means of the pistons. In addition, the controller may be configured to allow independent movement of the pistons and hence the rotors for other purposes, for example rotor cleaning.

Figure 8 illustrates a second embodiment of a screw pump having active control of the positions of the rotors within the stator. Similar to the first embodiment, the pump 70 includes a pump body 72 defining a pumping chamber 74, fluid inlet 76 and fluid outlet 78. The pump 70 further includes a first shaft 80 and, spaced therefrom and parallel thereto, a second shaft 82. First bearing assemblies 84 are provided for supporting the upper ends (as shown in Figure 8) of the shafts 80, 82, and second bearing assemblies 86 located within bearing housing 88 are provided for supporting the lower ends of the shafts 80, 82. The shafts 80, 82 are adapted for rotation within gearbox 83 about the longitudinal axes in a contra-rotational direction. One of the shafts 80 is connected to a drive motor (not shown) via a drive mechanism, the shafts being coupled together by means of timing gears 90 so that in use the shafts 80, 82 rotate at the same speed but in opposite directions.

A first rotor 92 is mounted on the first shaft 80 for rotary movement within the chamber 74, and a second rotor 94 is similarly mounted on the second shaft 82. Each of the two rotors 92, 94 has a first part proximate the inlet 76 having a generally cylindrical shape and a second part proximate the outlet 78 having a tapered shape. Each rotor has a helical vane or thread 96, 98 respectively formed on the outer surface thereof, the threads intermeshing as illustrated.

The screw threads 96, 98 of the rotors 92, 94 have, on the first part of each rotor, a substantially constant outer diameter and, on the second part of each rotor, an outside diameter which tapers decreasingly towards the outlet 78 of the pump 70. The inner surface of the pumping chamber 74, which acts as a stator during use of the pump, is conformingly shaped to the shape of the outer diameters of the rotors.

In order to control the axial position of the rotors 92, 94 within the pumping chamber 74, the pump 70 includes a servo motor 100 which rotates a lead screw 102 attached thereto. The lead screw 102 engages a conformingly-threaded aperture 104 in the bearing housing 88 so that the bearing housing 88 acts a piston, moving axially relative to the pumping chamber 74 to control the rotor to stator clearance over the tapered section of the pump 70. Actuation of the servo motor 100 can be controlled by any suitable mechanism. For example, a sensor 106, for example a Hall effect sensor 106 can provide to the motor 100, or, as illustrated to a separate controller 108 thereof, a signal indicative of the size *d*, or the rate of change, of the axial clearance between the rotors and the stator, operation of the motor 100 being controlled in accordance with the signal received from the sensor 106 to axially move the rotors to increase or decrease the size of the axial clearance as required.

The mechanism for axially moving the rotors relative to the stator in this second embodiment can, of course, be used to move rotors with wholly tapered screw-threads, as used in the first embodiment. In the third embodiment shown in Figure 9, this mechanism is used to axially move rotors 110 with non-tapered screw threads 112, and thereby control the axial clearance between the rotors 110 and the end surfaces 116, 118 of the stator 114, for example, to scrape process media from the ends of the rotors 110 and to prevent restart failure. In this embodiment, the fixed bearing assemblies 84 have been replaced by a floating bearing assembly, in which the bearings are located within bearing housing 120 which moves axially with axial movement of the rotors 110. Close tolerances between the outer walls 122 of the bearing housing 120 and the walls 124 of the

gearbox 83 serve to control the radial position of the bearing housing 120, the outer walls 122 of the bearing housing 120 carrying a fluid sealing mechanism (not shown) to prevent oil from the gearbox 83 entering the pumping chamber 126. The housing 88 of the bearing assemblies 86 carries a similar sealing mechanism (not shown) to seal the pumping chamber 126.

In the embodiment shown in Figure 10, the screw-threaded rotors 110 have been replaced by Northey rotors 130 to provide a multi-stage positive displacement pump, with axial movement of the rotors controlling the axial clearance between the faces of the rotors 130 and the opposing surfaces of the stator 132.

Figure 11 illustrates a dry pump having Roots rotors 140 which, similar to the screw threads in the first embodiment, taper decreasingly in a direction from the inlet to the outlet of the pump, the inner surface of the stator 142 conformingly tapering towards the pump outlet. In this embodiment, axial movement of the rotors 140 controls the radial clearance between the rotor 140 and the stator 142. In this embodiment, the axial clearances between the ends of the rotors 140 and the end surfaces 144, 146 of the pumping chamber housing the rotors 140 are maintained substantially constant with axial movement of the rotors relative to the stator 142. As shown in Figure 11, the end surfaces 144, 146 are defined by the housings 88, 120 of the bearing assemblies supporting the ends of the shafts 80, 82 so that the end surfaces 144, 146 move with the rotors 140.

In Figures 8 to 11, the mechanisms for axially moving the rotors relative to the stator are located at the low pressure (inlet) end of the pump. However, this mechanism could alternatively be located at the high pressure (exhaust) end of the pump.

Furthermore, whilst in the embodiments shown in Figures 8 to 11, the axial clearance is monitored by the sensor 106, other operational parameters of the pump, such as back pressure, exhaust temperature, power consumption and/or inlet pressure, can be conveyed to the controller 108, or direct to the motor 100,

for use in controlling the axial position of the rotors during use of the pump 70. Such signals may be output from suitably located sensors, or from the motor driving rotor rotation, as appropriate,

As well as controlling the axial position of the rotors in response to signals output from such sensors, other operational parameters can be adjusted in response to these signals. For example, the temperature and/or flow rate of cooling water supplied to a cooling jacket or other heat transfer device extending about the stator can be adjusted depending on the size of the clearance, enabling the stator to track the expansion and contraction of the rotors.

In summary, there is provided a dry pump comprising a stator housing first and second intermeshing rotors adapted for counter-rotation within the stator, and means for actively controlling the axial position of the rotors within the stator during use of the pump.